

A New Approach to Active Vibration Isolation for Microgravity Space Experiments

Alok Sinha and Chikuan K. Kao
The Pennsylvania State University
University Park, Pennsylvania

and

Carlos M. Grodsinsky
Lewis Research Center
Cleveland, Ohio

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A NEW APPROACH TO ACTIVE VIBRATION ISOLATION FOR MICROGRAVITY

SPACE EXPERIMENTS

Alok Sinha and Chikuan K. Kao*
Department of Mechanical Engineering
The Pennsylvania State University
University Park, Pennsylvania 16802

and

Carlos M. Grodsinsky
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

SUMMARY

E-5257 A new method has been developed to design an active vibration isolation system for microgravity space experiments. This method yields the required controller transfer functions for a specified transmissibility ratio. Hence, it is a straightforward task to guarantee that the desired vibration isolation performance is achieved at each frequency. The theory for such a controller design has been presented by considering a single degree of freedom system. In addition, the magnitude of the input required by the new method has been found to be less than that used by a standard phase lead/lag compensator.

1. INTRODUCTION

A number of experiments in the areas of fluid and material sciences are currently being designed to be performed on the STS orbiter, and in the future, on a space station in order to utilize the low gravity environment. However, the acceleration levels demanded by scientists for these experiments are well below what have been found on the man-tended spacecraft. The sources of relatively high level of acceleration include modal response of the space craft, crew motion, on-board rotating equipment etc. as well as some moderate to high amplitude accelerations from aerodynamic drag, gravity gradient and photon pressure (ref. 1). Jones et al. (refs. 2 and 3) have presented the frequency spectrum of the acceleration levels found on a space craft, figure 1(a). This frequency spectrum, which has been described as a conservative estimate, has an f^2 variation below 3 Hz and a constant level of 10^{-1} g above 3 Hz. The frequency spectrum of the acceleration level required for many experiments is shown in figure 1(b). The dotted line represents the proposed requirement for the US lab modules. On the basis of figures 1(a) and (b), the required acceleration transmissibility ratio for a vibration isolation system has been obtained, figure 1(c). Below 0.03 Hz (or 0.01 Hz),¹ the experimental module is allowed to follow the space station acceleration, and above 0.03 Hz (or 0.01 Hz), a -40 dB/decade roll-off is required. In terms of Bode plot's terminology (ref. 4), the isolation system should have a break frequency of 0.03 Hz

*Presently at General Motors Research Laboratories, Warren, Michigan.

¹Note that 0.01 Hz corresponds to the dotted line in figure 1(c).

(or 0.01 Hz). The desired vibration isolation performance can be realized by achieving a transfer function of the form $1/(\tau S + 1)^2$ where $1/\tau = 0.03$ Hz (or 0.01 Hz).

Recently, Garriott and DeBra (ref. 5) have proposed a passive isolation system, which utilizes a set of mechanical springs. However, the natural frequency of the system must be chosen to be 0.03 Hz (or 0.01 Hz) in order to have the break frequency equal 0.03 Hz (or 0.01 Hz). The requirement for such a low natural frequency implies an extremely low stiffness for typical values of the mass found in space experiment packages. Furthermore, it is well-known (ref. 6) that the presence of any damping has a detrimental effect on the performance of a passive vibration isolation system. Hence, the damping has to be chosen to equal zero. In this case, the transfer function for the vibration isolation system would be $1/(\tau^2 S^2 + 1)$ where $1/\tau = 0.03$ Hz. Consequently, the passive system would yield a -40 dB/decade roll-off only in an asymptotic sense; and would have an extremely poor performance near the break frequency due to resonance.

Jones et al. (refs. 2 and 3) have developed an active microgravity isolation mount. They used a Lorentz type electromagnetic actuator (ref. 7) and utilized the relative displacement of the mass to its dynamic support as the reference signal for a phase lead/lag compensated control loop. Although they have demonstrated that it is feasible to develop hardware for the design of an active isolation mount, their controller is of phase lead/lag type (ref. 4). In the context of a single degree of freedom system, the resulting transfer function, which represents the transmissibility ratio, has a first order numerator and a third order denominator; i.e., there are one zero and three poles. By tuning the controller parameters, the transmissibility ratio can be forced to be close to the desired function. However, it is not a straightforward task to tune the controller parameters and, in many cases, the transmissibility ratio has been found to exceed its desired value (refs. 2 and 3).

In this paper, a new approach has been developed to design an active vibration isolation system. The relative displacement and the acceleration of the mass, which can be measured directly or indirectly on the space craft, are used as feedback signals. A methodology has been developed to determine the controller transfer function for a specified transmissibility ratio function. Hence, it is straightforward to guarantee that the transmissibility ratio is below its upper bound at each frequency. The theory for the controller design has been presented in the context of a single degree of freedom system, figure 2. The extension of this theory to a multidegree of freedom system will be the topic of subsequent papers.

First, the open-loop system is described. Then, the theory for the new controller design is developed and illustrated using a number of examples. Lastly, the input required for the new approach is compared to that for the phase lead/lag compensator.

2. OPEN-LOOP SYSTEM

The single degree of freedom spring-mass system is shown in figure 2. The parameters m and k represent the mass of the experiment module and the umbilical stiffness, respectively. The stiffness k is due to the connections between the experiment module and the base, which are necessary for various

functions; e.g., the supply for electric power, the transport of cooling fluid, etc. The base acceleration \ddot{x} represents the acceleration of the space station.

The dynamics of the system shown in figure 2 is represented by

$$m\ddot{y} + k(y - x) = 0 \quad (1)$$

The transfer function relating $Y(s)$ and $X(s)$ is as follows:

$$\frac{Y(S)}{X(S)} = \frac{k}{mS^2 + k} \quad (2)$$

3. NEW APPROACH FOR THE CONTROLLER DESIGN

On the STS orbiter or a space station, the absolute displacement of the mass cannot be directly measured. However, the relative displacement $(y-x)$ and the absolute acceleration (\ddot{y}) can be measured. Therefore, these signals are used to develop the new controller design. The block diagram for this control system is shown in figure 3, where $H_1(S)$ and $H_2(S)$ are the controller transfer functions corresponding to the acceleration and the relative displacement feedback, respectively. The transfer function relating $Y(S)$ and $X(S)$, which is the transmissibility ratio, is as follows:

$$\frac{Y(S)}{X(S)} = \frac{k + k_1 H_2(S)}{S^2(m + k_1 H_1(S)) + k + k_1 H_2(S)} \quad (3)$$

Let $H_d(S)$ be the desired transfer function; i.e., one wants to achieve

$$\frac{Y(S)}{X(S)} = H_d(S) \quad (4)$$

From equations (3) and (4),

$$\frac{k + k_1 H_2(S)}{S^2(m + k_1 H_1(S)) + k + k_1 H_2(S)} = H_d(S) \quad (5)$$

Equation (5) has two unknowns: $H_1(S)$ and $H_2(S)$. Therefore, either $H_1(S)$ or $H_2(S)$ can be arbitrarily chosen. For a specified $H_1(S)$,

$$H_2(S) = \frac{[S^2(m + k_1 H_1(S)) + k]H_d(S) - k}{k_1(1 - H_d(S))} \quad (6)$$

Similarly, for a specified $H_2(S)$,

$$H_1(S) = \frac{(k + k_1 H_2(S))(1 - H_d(S)) - S^2 m H_d(S)}{S^2 k_1 H_d(S)} \quad (7)$$

Examples:

(1) If $H_1(S) = 0$ and $H_d(S) = 1/(\tau S + 1)^2$,

$$H_2(S) = \frac{(m - k\tau^2)(S - 2\tau k/(m - k\tau^2))}{k_1\tau^2(S + 2/\tau)} \quad (8)$$

It should be noted that only the relative displacement is fed back in this example, and the controller transfer function is of phase lead/lag type.

(2) If $H_1(S) = 0$ and $H_d(S) = 1/(\tau S + 1)^3$,

$$H_2(S) = \frac{-k\tau^3 S^2 + (m - 3k\tau^2)S - 3k\tau}{k_1\tau(\tau^2 S^2 + 3\tau S + 3)} \quad (9)$$

(3) If $H_2(S) = 0$ and $H_d(S) = 1/(\tau S + 1)^2$,

$$H_1(S) = k(1/H_d(S) - 1)/(k_1 S^2) - m/k_1 \quad (10)$$

3.1 Simultaneous Choice of $H_1(S)$ and $H_2(S)$

To determine $H_1(S)$ and $H_2(S)$ uniquely, an additional constraint is sought by examining their effects on the required input magnitude. For a desired transfer function $H_d(S)$, the control input $U(S)$ is related to the base displacement $X(S)$ as follows:

$$U(S) = [(mS^2 + k)H_d(S) - k]X(S) \quad (11)$$

This equation indicates that the control input (or the actuator's size) is primarily determined by $H_d(S)$ and is not dependent on $H_1(S)$ and $H_2(S)$. However, the additional constraint can be obtained by specifying the fraction of the input that comes from each feedback loop shown in figure 3. For example, if both feedback loops are required to contribute equally to the input,

$$U_1(S) = U_2(S) \quad (12)$$

From figure 3,

$$U_1(S) = H_1(S)S^2H_d(S)X(S) \quad (13)$$

$$U_2(S) = H_2(S)[H_d(S) - 1]X(S) \quad (14)$$

From equations (12) to (14),

$$H_1(S)S^2H_d(S) = H_2(S)[H_d(S) - 1] \quad (15)$$

Solving equations (5) and (15) simultaneously,

$$H_1(S) = \frac{k(1 - H_d(S)) - mS^2 H_d(S)}{2k_1 S^2 H_d(S)} \quad (16)$$

and

$$H_2(S) = \frac{k(1 - H_d(S)) - mS^2 H_d(S)}{2k_1 [H_d(S) - 1]} \quad (17)$$

Example: If $H_d(S) = 1/(\tau S + 1)^2$, equations (16) and (17) yield

$$H_1(S) = \frac{k\tau \left(\frac{k\tau^2}{2k\tau} S + 1 \right)}{k_1 S} \quad (18)$$

and

$$H_2(S) = \frac{-k \left(\frac{k\tau^2}{2k\tau} S + 1 \right)}{k_1 (\tau S + 2)} \quad (19)$$

4. COMPARISON WITH THE PHASE LEAD/LAG COMPENSATOR METHOD

The method proposed in Section 3 yields the desired transmissibility ratio for all values of m and k . In this respect, it is obvious that the new method is superior to the lead/lag compensator method. However, it would be important to compare the required input magnitudes for both cases. For the sake of comparison, only the following part of the input in equation (11) needs to be considered:

$$U_c(S) = (mS^2 + k)G(S)X(S) \quad (20)$$

where

$$G(S) = \frac{Y(S)}{X(S)} \quad (21)$$

From equation (20),

$$G_u(S) = U_c(S)/X(S) = (mS^2 + k)G(S) \quad (22)$$

Case I: Phase Lead/Lag Compensator Method (refs. 2 and 3)

In this case, the controller has the following structures:

$$H_1(S) = 0 \text{ and } H_2(S) = \frac{C(S + b)}{(S + a)} \quad (23)$$

The resulting transmissibility ratio is as follows:

$$G(S) = \frac{S_1 + \gamma}{S_1^3 + \alpha S_1^2 + S_1 + \gamma} \quad (24)$$

where

$$S_1 = S/\omega_0$$

$$\omega_0^2 = (k + k_1 C)/m$$

$$\alpha = a/\omega_0$$

$$\gamma = \alpha(k + bk_1 C/a)/(k + k_1 C)$$

For $m = 100$ kg, Jones et al. (refs. 2 and 3) used $\alpha = 1.25$ and $\gamma = \alpha/30$. In this case, the resulting values of a , b and $k_1 C$ are shown in table I for three values of k : 0, 1 and 5 N/m. In figure 4, the frequency response $G(j\omega)$, is plotted using equation (24). Although this transmissibility ratio is close to the desired performance (fig. 1(c)), it does exceed the desired value near 0.03 Hz. In figures 5 to 7, $|G_U(j\omega)|$ has been plotted for the three values of umbilical stiffness k .

Case II: New Method With Only Relative Displacement Feedback

The controller structure is given by equation (8) for $G(S) = 1/(\tau S + 1)^2$. Hence, the form of the controller can also be described by equation (23). In table I, the values of a , b and C are given for the three different values of k , where $1/\tau = 0.03$ Hz. In this case, the transmissibility ratio is below the desired value at each frequency, figure 4. In figures 5 to 7, the frequency spectrums of $G_U(S)$ are shown.

The results in figures 4 to 7 show that the proposed method satisfies the performance requirements at all frequencies; and also requires an input smaller than that used by a phase lead/lag compensator. Hence, the actuators developed by Jones et al. (refs. 2 and 3) can be used to implement the new controller.

Lastly, it should be noted that Jones et al. (refs. 2 and 3) have used $\alpha = 0.5$ and 2.5 for $m = 50$ and 200 kg, respectively. In these cases, the transmissibility ratio exceeds the desired function by an amount greater than that shown in figure 4 for $m = 100$ kg. Hence, the performance of the new controller has been compared to the best result that has been obtained by the phase lead/lag method.

5. CONCLUSIONS

A new method has been developed to design an active vibration isolation system for microgravity science experiments. The relative displacement and the acceleration of the isolated mass has been used as feedback signals. Using the method presented in this paper, it is a straightforward process to determine the controller transfer functions $H_1(S)$ and $H_2(S)$ (fig. 3), which will guarantee that the resulting transmissibility ratio will be below its specified upper bound at each frequency. The required input is found to be independent of $H_1(S)$ and $H_2(S)$; and solely depends on the desired transmissibility ratio $H_d(S)$. The input required by the controller designed with only the relative displacement feedback and $H_d(S) = 1/(\tau S + 1)^2$ is found to be smaller than that required by the phase lead/lag approach. Hence, the actuators developed by Jones et al. (refs. 2 and 3) could be directly used to implement the new controller.

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TABLE I. - CONTROLLER PARAMETERS

Case	k = 0			k = 1 N/m			k = 5 N/m		
	a	b	k_1C	a	b	k_1C	a	b	k_1C
I	0.236	7.854×10^{-3}	3.553	0.236	-0.0814	2.533	0.236	0.796	-1.447
II	.377	0	3.553	.377	-.148	2.533	.377	1.303	-1.447

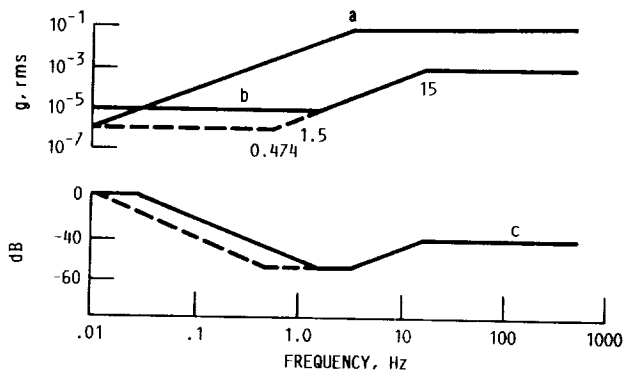


FIGURE 1. - (a) MAGNITUDE OF ACCELERATION ON SPACE STATION [2,3]; (b) REQUIRED MAGNITUDE OF ACCELERATION [2,3]; (c) DESIRED TRANSMISSIBILITY RATIO [2,3].

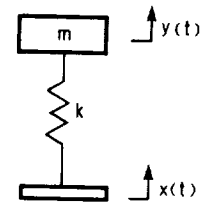


FIGURE 2. - OPEN LOOP SYSTEM.

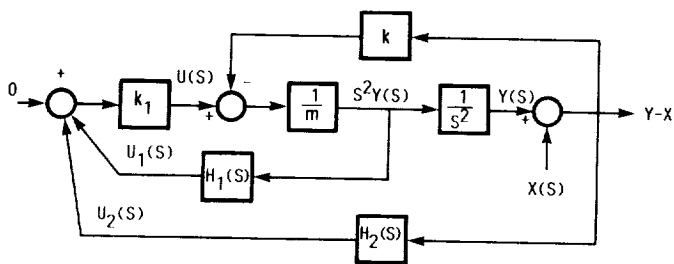


FIGURE 3. - STRUCTURE OF THE CONTROL SYSTEM.

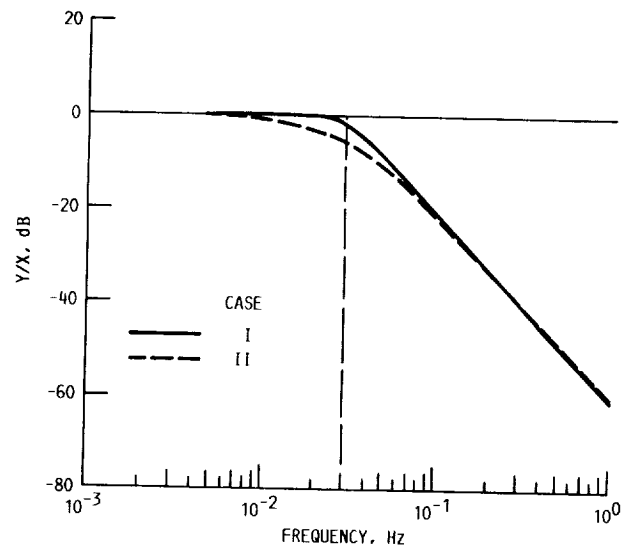


FIGURE 4. - TRANSMISSIBILITY RATIOS.

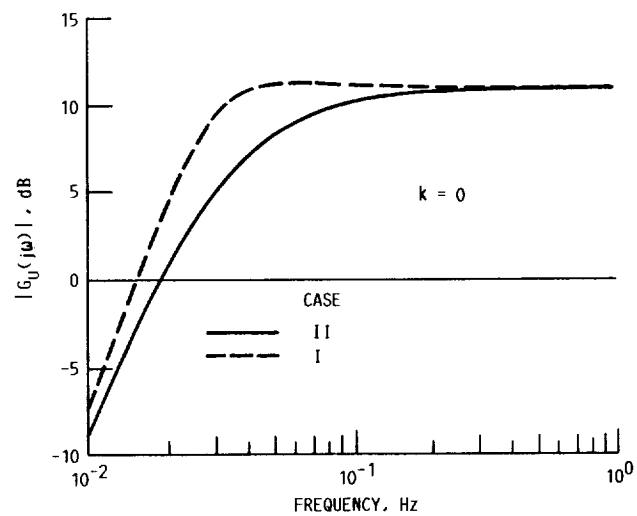


FIGURE 5. - FREQUENCY SPECTRUM OF $G_U(s)$, $k = 0$.

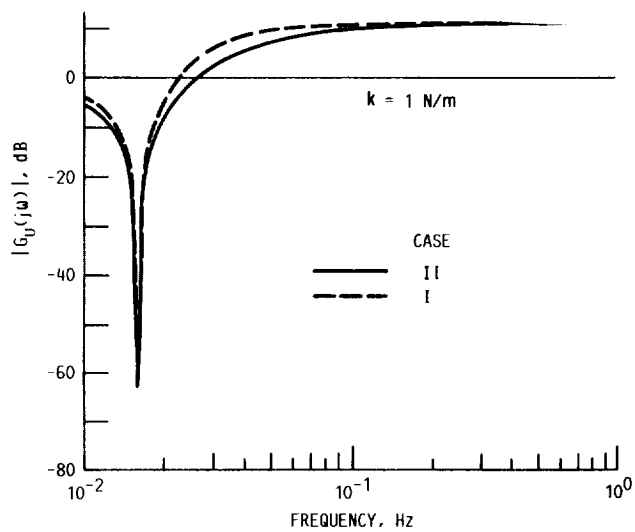


FIGURE 6. - FREQUENCY SPECTRUM OF $G_U(s)$, $k = 1 \text{ N/m}$.

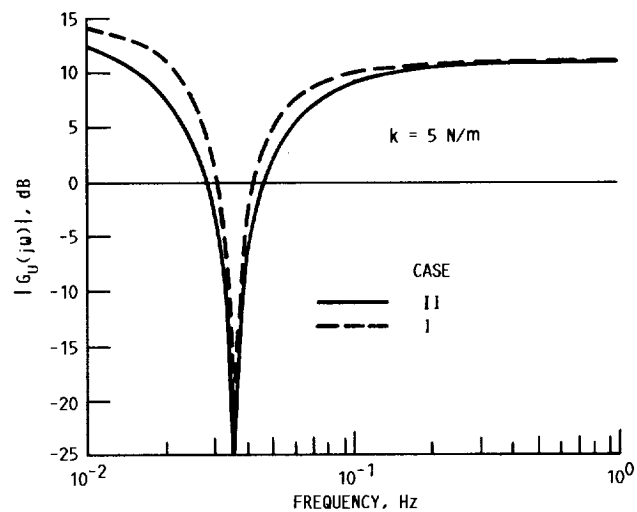


FIGURE 7. - FREQUENCY SPECTRUM OF $G_U(s)$, $k = 5 \text{ N/m}$.



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